Properties of Lubricating Oil

Density. This property has no relation to lubricating value but is useful in changing the kinematic viscosity to absolute viscosity. Mathematically

$$\text{Absolute viscosity} = \rho \times \text{Kinematic viscosity (in } m^2/s)$$

where ρ = Density of the lubricating oil.

The density of most of the oils at 15.5°C varies from 860 to 950 kg / m^3 (the average value may be taken as 900 kg / m^3). The density at any other temperature (t) may be obtained from the following relation, *i.e.*

$$\rho_t = \rho_{15.5} - 0.000\ 657\ t$$

where $\rho_{15.5}$ = Density of oil at 15.5° C.

Viscosity index. The term viscosity index is used to denote the degree of variation of viscosity

Flash point. It is the lowest temperature at which an oil gives off sufficient vapour to support a momentary flash without actually setting fire to the oil when a flame is brought within 6 mm at the surface of the oil.

Fire point. It is the temperature at which an oil gives off sufficient vapour to burn it

. Pour point or freezing point. It is the temperature at which an oil will cease to flow when cooled.

Terms used in Hydrodynamic Journal Bearing

A hydrodynamic journal bearing is shown in Fig. 26.7, in which O is the centre of the journal and O' is the centre of the bearing.

Let D = Diameter of the bearing,

d = Diameter of the journal, and

l = Length of the bearing.

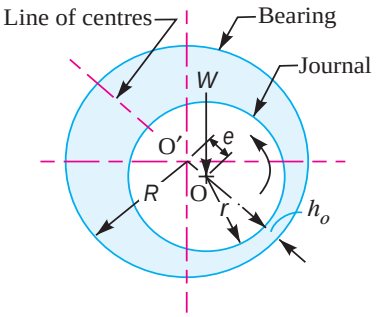

Fig. 26.7. Hydrodynamic journal bearing.

The following terms used in hydrodynamic journal bearing are important from the subject point of view :

1. *Diametral clearance.* It the difference between the diameters of the bearing and the journal. Mathematically, diametral clearance,

$$c = D - d$$

Note : The diametral clearance (c) in a bearing should be small enough to produce the necessary velocity gradient, so that the pressure built up will support the load. Also the small clearance has the advantage of decreasing side leakage. However, the allowance must be made for manufacturing tolerances in the journal and bushing. A commonly used clearance in industrial machines is 0.025 mm per cm of journal diameter.

2. *Radial clearance.* It is the difference between the radii of the bearing and the journal. Mathematically, radial clearance,

$$c_1 = R - r = \frac{D - d}{2} = \frac{c}{2}$$

3. *Diametral clearance ratio.* It is the ratio of the diametral clearance to the diameter of the journal. Mathematically, diametral clearance ratio

$$= \frac{c}{d} = \frac{D - d}{d}$$

4. *Eccentricity.* It is the radial distance between the centre (O) of the bearing and the displaced centre (O') of the bearing under load. It is denoted by e.

5. *Minimum oil film thickness.* It is the minimum distance between the bearing and the journal, under complete lubrication condition. It is denoted by h_0 and occurs at the line of centres as shown in Fig. 26.7. Its value may be assumed as $c / 4$.

6. *Attitude or eccentricity ratio.* It is the ratio of the eccentricity to the radial clearance. Mathematically, attitude or eccentricity ratio,

$$\varepsilon = \frac{e}{c_1} = \frac{c_1 - h_0}{c_1} = 1 - \frac{h_0}{c_1} = 1 - \frac{2h_0}{c} \quad \ldots (\because c_1 = c / 2)$$

7. *Short and long bearing.* If the ratio of the length to the diameter of the journal (*i.e.* l / d) is less than 1, then the bearing is said to be **short bearing.** On the other hand, if l / d is greater than 1, then the bearing is known as ***long bearing.***

Axle bearings

Notes : 1. When the length of the journal (l) is equal to the diameter of the journal (d), then the bearing is called ***square bearing.***

2. Because of the side leakage of the lubricant from the bearing, the pressure in the film is atmospheric at the ends of the bearing. The average pressure will be higher for a long bearing than for a short or square bearing. Therefore, from the stand point of side leakage, a bearing with a large l / d ratio is preferable. However, space requirements, manufacturing, tolerances and shaft deflections are better met with a short bearing. The value of l / d may be taken as 1 to 2 for general industrial machinery. In crank shaft bearings, the l / d ratio is frequently less than 1.

Bearing Characteristic Number and Bearing Modulus for Journal Bearings

The coefficient of friction in design of bearings is of great importance, because it affords a means for determining the loss of power due to bearing friction. It has been shown by experiments that the coefficient of friction for a full lubricated journal bearing is a function of three variables, *i.e.*

(*i*) $\frac{ZN}{p}$; **(*ii*)** $\frac{d}{c}$; and **(*iii*)** $\frac{l}{d}$

Therefore the coefficient of friction may be expressed as

$$\mu = \phi\left(\frac{ZN}{p}, \frac{d}{c}, \frac{l}{d}\right)$$

where μ = Coefficient of friction,

ϕ = A functional relationship,

Z = Absolute viscosity of the lubricant, in kg / m-s,

N = Speed of the journal in r.p.m.,

p = Bearing pressure on the projected bearing area in N/mm^2,

= Load on the journal $\div$ $l \times d$

d = Diameter of the journal,

l = Length of the bearing, and

c = Diametral clearance.

The factor ZN / p is termed as ***bearing characteristic number*** and is a dimensionless number. The variation of coefficient of friction with the operating values of bearing characteristic number (ZN / p) as obtained by McKee brothers (S.A. McKee and T.R. McKee) in an actual test of friction is shown in Fig. 26.8. The factor ZN/p helps to predict the performance of a bearing.

The part of the curve *PQ* represents the region of thick film lubrication. Between *Q* and *R*, the viscosity (*Z*) or the speed (*N*) are so low, or the pressure (*p*) is so great that their combination *ZN / p* will reduce the film thickness so that partial metal to metal contact will result. The thin film or boundary lubrication or imperfect lubrication exists between *R* and *S* on the curve. This is the region where the viscosity of the lubricant ceases to be a measure of friction characteristics but the oiliness of the lubricant is effective in preventing complete metal to metal contact and seizure of the parts.

Clutch bearing

It may be noted that the part *PQ* of the curve represents stable operating conditions, since from any point of stability, a decrease in viscosity (*Z*) will reduce *ZN / p*. This will result in a decrease in coefficient of friction (μ) followed by a lowering of bearing temperature that will raise the viscosity (*Z*).

From Fig. 26.8, we see that the minimum amount of friction occurs at *A* and at this point the value of *ZN / p* is known as ***bearing modulus*** which is denoted by *K*. The bearing should not be operated at this value of bearing modulus, because a slight decrease in speed or slight increase in pressure will break the oil film and make the journal to operate with metal to metal contact. This will result in high friction, wear and heating. In order to prevent such conditions, the bearing should be designed for a value of *ZN / p* at least three times the minimum value of bearing modulus (*K*). If the bearing is subjected to large fluctuations of load and heavy impacts, the value of *ZN / p* = 15 *K* may be used.

From above, it is concluded that when the value of *ZN / p* is greater than *K*, then the bearing will operate with thick film lubrication or under hydrodynamic conditions. On the other hand, when the value of *ZN / p* is less than *K*, then the oil film will rupture and there is a metal to metal contact.

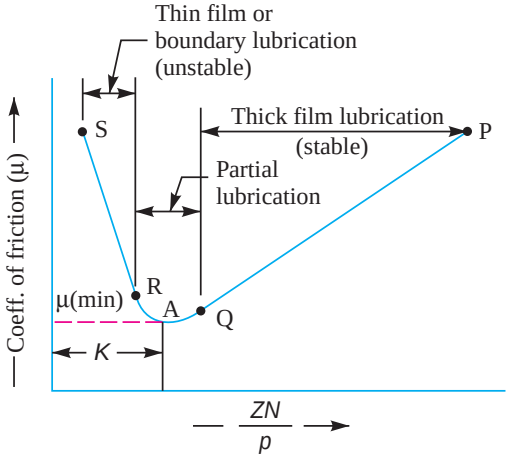

Fig. 26.8. Variation of coefficient of friction with *ZN*/*p*.

Coefficient of Friction for Journal Bearings

In order to determine the coefficient of friction for well lubricated full journal bearings, the following empirical relation established by McKee based on the experimental data, may be used.

*Coefficient of friction,

$$\mu = \frac{33}{10^8}\left(\frac{ZN}{p}\right)\left(\frac{d}{c}\right) + k \quad \text{... (when Z is in kg / m-s and } p \text{ is in N / mm}^2\text{)}$$

where Z, N, p, d and c have usual meanings as discussed in previous article, and

k = Factor to correct for end leakage. It depends upon the ratio of length to the diameter of the bearing (*i.e.* l / d).

= 0.002 for l / d ratios of 0.75 to 2.8.

The operating values of ZN / p should be compared with values given in Table 26.3 to ensure safe margin between operating conditions and the point of film breakdown.

Table 26.2. Absolute viscosity of commonly used lubricating oils.

S. No.	Type of oil	Absolute viscosity in kg / m-s at temperature in °C											
		30	35	40	45	50	55	60	65	70	75	80	90
1.	SAE 10	0.05	0.036	0.027	0.0245	0.021	0.017	0.014	0.012	0.011	0.009	0.008	0.005
2.	SAE 20	0.069	0.055	0.042	0.034	0.027	0.023	0.020	0.017	0.014	0.011	0.010	0.0075
3.	SAE 30	0.13	0.10	0.078	0.057	0.048	0.040	0.034	0.027	0.022	0.019	0.016	0.010
4.	SAE 40	0.21	0.17	0.12	0.096	0.78	0.06	0.046	0.04	0.034	0.027	0.022	0.013
5.	SAE 50	0.30	0.25	0.20	0.17	0.12	0.09	0.076	0.06	0.05	0.038	0.034	0.020
6.	SAE 60	0.45	0.32	0.27	0.20	0.16	0.12	0.09	0.072	0.057	0.046	0.040	0.025
7.	SAE 70	1.0	0.69	0.45	0.31	0.21	0.165	0.12	0.087	0.067	0.052	0.043	0.033

Table 26.3. Design values for journal bearings.

Machinery	*Bearing*	*Maximum bearing pressure (p) in N/mm²*	*Operating values*			
			Absolute Viscosity (Z) in kg/m-s	*ZN/p Z in kg/m-s p in N/mm²*	$\frac{c}{d}$	$\frac{l}{d}$
Automobile and air-craft engines	Main	5.6 – 12	0.007	2.1	—	0.8 – 1.8
	Crank pin	10.5 – 24.5	0.008	1.4		0.7 – 1.4
	Wrist pin	16 – 35	0.008	1.12		1.5 – 2.2
Four stroke-Gas and oil engines	Main	5 – 8.5	0.02	2.8	0.001	0.6 – 2
	Crank pin	9.8 – 12.6	0.04	1.4		0.6 – 1.5
	Wrist pin	12.6 – 15.4	0.065	0.7		1.5 – 2
Two stroke-Gas and oil engines	Main	3.5 – 5.6	0.02	3.5	0.001	0.6 – 2
	Crank pin	7 – 10.5	0.04	1.8		0.6 – 1.5
	Wrist pin	8.4 – 12.6	0.065	1.4		1.5 – 2
Marine steam engines	Main	3.5	0.03	2.8	0.001	0.7 – 1.5
	Crank pin	4.2	0.04	2.1		0.7 – 1.2
	Wrist pin	10.5	0.05	1.4		1.2 – 1.7
Stationary, slow speed steam engines	Main	2.8	0.06	2.8	0.001	1 – 2
	Crank pin	10.5	0.08	0.84		0.9 – 1.3
	Wrist pin	12.6	0.06	0.7		1.2 – 1.5
Stationary, high speed steam engine	Main	1.75	0.015	3.5	0.001	1.5 – 3
	Crank pin	4.2	0.030	0.84		0.9 – 1.5
	Wrist pin	12.6	0.025	0.7		13 – 1.7
Reciprocating pumps and compressors	Main	1.75	0.03	4.2	0.001	1 – 2.2
	Crank pin	4.2	0.05	2.8		0.9 – 1.7
	Wrist pin	7.0	0.08	1.4		1.5 – 2.0
Steam locomotives	Driving axle	3.85	0.10	4.2	0.001	1.6 – 1.8
	Crank pin	14	0.04	0.7		0.7 – 1.1
	Wrist pin	28	0.03	0.7		0.8 – 1.3

* This is the equation of a straight line portion in the region of thick film lubrication (*i.e.* line *PQ*) as shown in Fig. 26.8.

Machinery	*Bearing*	*Maximum bearing pressure (p) in N/mm²*	*Operating values*			
			Absolute Viscosity (Z) in kg/m-s	*ZN/p Z in kg/m-s p in N/mm²*	$\frac{c}{d}$	$\frac{l}{d}$
Railway cars	Axle	3.5	0.1	7	0.001	1.8 – 2
Steam turbines	Main	0.7 – 2	0.002 – 0.016	14	0.001	1 – 2
Generators, motors, centrifugal pumps	Rotor	0.7 – 1.4	0.025	28	0.0013	1 – 2
Transmission shafts	Light, fixed	0.175	0.025-	7	0.001	2 – 3
	Self -aligning	1.05	0.060	2.1		2.5 – 4
	Heavy	1.05		2.1		2– 3
Machine tools	Main	2.1	0.04	0.14	0.001	1–4
Punching and shearing machines	Main	28	0.10	—	0.001	1–2
	Crank pin	56				
Rolling Mills	Main	21	0.05	1.4	0.0015	1–1.5

Critical Pressure of the Journal Bearing

The pressure at which the oil film breaks down so that metal to metal contact begins, is known as **critical pressure** or the **minimum operating pressure** of the bearing. It may be obtained by the following empirical relation, *i.e.*

Critical pressure or minimum operating pressure,

$$p = \frac{ZN}{4.75 \times 10^6} \left(\frac{d}{c}\right)^2 \left(\frac{l}{d + l}\right) \text{N/mm}^2 \qquad \text{...(when Z is in kg / m-s)}$$

Sommerfeld Number

The Sommerfeld number is also a dimensionless parameter used extensively in the design of journal bearings. Mathematically,

$$\text{Sommerfeld number} = \frac{ZN}{p}\left(\frac{d}{c}\right)^2$$

For design purposes, its value is taken as follows :

$$\frac{ZN}{p}\left(\frac{d}{c}\right)^2 = 14.3 \times 10^6 \qquad \text{... (when } Z \text{ is in kg / m-s and } p \text{ is in N / mm}^2\text{)}$$

Heat Generated in a Journal Bearing

The heat generated in a bearing is due to the fluid friction and friction of the parts having relative motion. Mathematically, heat generated in a bearing,

$$Q_g = \mu.W.V \text{ N-m/s or J/s or watts} \qquad \text{...(i)}$$

where μ = Coefficient of friction,

W = Load on the bearing in N,

= Pressure on the bearing in N/mm^2 × Projected area of the bearing in mm^2 = p ($l \times d$),

V = Rubbing velocity in m/s = $\frac{\pi d.N}{60}$, d is in metres, and

N = Speed of the journal in r.p.m.

After the thermal equilibrium has been reached, heat will be dissipated at the outer surface of the bearing at the same rate at which it is generated in the oil film. The amount of heat dissipated will depend upon the temperature difference, size and mass of the radiating surface and on the amount of air flowing around the bearing. However, for the convenience in bearing design, the actual heat dissipating area may be expressed in terms of the projected area of the journal.

Heat dissipated by the bearing,

$$Q_d = C.A\ (t_b - t_a)\ \text{J/s or W} \qquad \text{... } (\because\ 1\ \text{J/s} = 1\ \text{W})\ \text{...}(ii)$$

where C = Heat dissipation coefficient in W/m^2/°C,

A = Projected area of the bearing in m^2 = $l \times d$,

t_b = Temperature of the bearing surface in °C, and

t_a = Temperature of the surrounding air in °C.

The value of C have been determined experimentally by O. Lasche. The values depend upon the type of bearing, its ventilation and the temperature difference. The average values of C (in W/m^2/°C), for journal bearings may be taken as follows :

For unventilated bearings (Still air)

= 140 to 420 W/m^2/°C

For well ventilated bearings

= 490 to 1400 W/m^2/°C

It has been shown by experiments that the temperature of the bearing (t_b) is approximately mid-way between the temperature of the oil film (t_0) and the temperature of the outside air (t_a). In other words,

$$t_b - t_a = \frac{1}{2}\ (t_0 - t_a)$$

Notes : 1. For well designed bearing, the temperature of the oil film should not be more than 60°C, otherwise the viscosity of the oil decreases rapidly and the operation of the bearing is found to suffer. The temperature of the oil film is often called as the ***operating temperature*** of the bearing.

2. In case the temperature of the oil film is higher, then the bearing is cooled by circulating water through coils built in the bearing.

3. The mass of the oil to remove the heat generated at the bearing may be obtained by equating the heat generated to the heat taken away by the oil. We know that the heat taken away by the oil,

$Q_t = m.S.t$ J/s or watts

where m = Mass of the oil in kg / s,

S = Specific heat of the oil. Its value may be taken as 1840 to 2100 J / kg / °C,

t = Difference between outlet and inlet temperature of the oil in °C.

Design Procedure for Journal Bearing

The following procedure may be adopted in designing journal bearings, when the bearing load, the diameter and the speed of the shaft are known.

1. Determine the bearing length by choosing a ratio of l / d from Table 26.3.
2. Check the bearing pressure, $p = W / l.d$ from Table 26.3 for probable satisfactory value.
3. Assume a lubricant from Table 26.2 and its operating temperature (t_0). This temperature should be between 26.5°C and 60°C with 82°C as a maximum for high temperature installations such as steam turbines.
4. Determine the operating value of ZN / p for the assumed bearing temperature and check this value with corresponding values in Table 26.3, to determine the possibility of maintaining fluid film operation.
5. Assume a clearance ratio c / d from Table 26.3.
6. Determine the coefficient of friction (μ) by using the McKee relation.

Journal bearings are used in helicopters, primarily in the main rotor axis and in the landing gear for fixed wing aircraft.

7. Determine the heat generated by using the relation as discussed previously.
8. Determine the heat dissipated by using the relation as discussed previously.
9. Determine the thermal equilibrium to see that the heat dissipated becomes atleast equal to the heat generated. In case the heat generated is more than the heat dissipated then either the bearing is redesigned or it is artificially cooled by water.

Example *Design a journal bearing for a centrifugal pump from the following data :*

Load on the journal = 20 000 N; Speed of the journal = 900 r.p.m.; Type of oil is SAE 10, for which the absolute viscosity at 55°C = 0.017 kg / m-s; Ambient temperature of oil = 15.5°C ; Maximum bearing pressure for the pump = 1.5 N / mm^2.

Calculate also mass of the lubricating oil required for artificial cooling, if rise of temperature of oil be limited to 10°C. Heat dissipation coefficient = 1232 W/m^2/°C.

Solution. Given : W = 20 000 N ; N = 900 r.p.m. ; t_0 = 55°C ; Z = 0.017 kg/m-s ; t_a = 15.5°C ; p = 1.5 N/mm^2 ; t = 10°C ; C = 1232 W/m^2/°C

The journal bearing is designed as discussed in the following steps :

1. First of all, let us find the length of the journal (l). Assume the diameter of the journal (d) as 100 mm. From Table 26.3, we find that the ratio of l / d for centrifugal pumps varies from 1 to 2. Let us take l / d = 1.6.

$$\therefore \qquad l = 1.6\, d = 1.6 \times 100 = 160 \text{ mm}$$ **Ans.**

2. We know that bearing pressure,

$$p = \frac{W}{l.d} = \frac{20\,000}{160 \times 100} = 1.25$$

Since the given bearing pressure for the pump is 1.5 N/mm^2, therefore the above value of p is safe and hence the dimensions of l and d are safe.

3.
$$\frac{Z.N}{p} = \frac{0.017 \times 900}{1.25} = 12.24$$

From Table 26.3, we find that the operating value of

$$\frac{Z.N}{p} = 28$$

We have discussed previously, that the minimum value of the bearing modulus at which the oil film will break is given by

$$3\ K = \frac{ZN}{p}$$

∴ Bearing modulus at the minimum point of friction,

$$K = \frac{1}{3}\left(\frac{Z.N}{p}\right) = \frac{1}{3} \times 28 = 9.33$$

Since the calculated value of bearing characteristic number $\left(\frac{Z.N}{p} = 12.24\right)$ is more than 9.33, therefore the bearing will operate under hydrodynamic conditions.

4. From Table 26.3, we find that for centrifugal pumps, the clearance ratio (c/d)

$$= 0.0013$$

5. We know that coefficient of friction,

$$\mu = \frac{33}{10^8}\left(\frac{ZN}{p}\right)\left(\frac{d}{c}\right) + k = \frac{33}{10^8} \times 12.24 \times \frac{1}{0.0013} + 0.002$$

$$= 0.0031 + 0.002 = 0.0051$$ ... [From Art. 26.13, $k = 0.002$]

6. Heat generated,

$$Q_g = \mu\ W\ V = \mu\ W \left(\frac{\pi d.N}{60}\right) \text{W} \quad \ldots\left(\because V = \frac{\pi d.N}{60}\right)$$

$$= 0.0051 \times 20\,000 \left(\frac{\pi \times 0.1 \times 900}{60}\right) = 480.7\ \text{W}$$

... (d is taken in metres)

7. Heat dissipated,

$$Q_d = C.A\ (t_b - t_a) = C.l.d\ (t_b - t_a)\ \text{W} \quad \ldots\ (\because A = l \times d)$$

We know that

$$(t_b - t_a) = \tfrac{1}{2}\ (t_0 - t_a) = \tfrac{1}{2}\ (55° - 15.5°) = 19.75°\text{C}$$

∴ $$Q_d = 1232 \times 0.16 \times 0.1 \times 19.75 = 389.3\ \text{W}$$

... (l and d are taken in metres)

We see that the heat generated is greater than the heat dissipated which indicates that the bearing is warming up. Therefore, either the bearing should be redesigned by taking $t_0 = 63°\text{C}$ or the bearing should be cooled artificially.

We know that the amount of artificial cooling required

$$= \text{Heat generated} - \text{Heat dissipated} = Q_g - Q_d$$

$$= 480.7 - 389.3 = 91.4\ \text{W}$$

Mass of lubricating oil required for artificial cooling

Let m = Mass of the lubricating oil required for artificial cooling in kg / s.

We know that the heat taken away by the oil,

$$Q_t = m.S.t = m \times 1900 \times 10 = 19\,000\ m\ \text{W}$$

... [∵ Specific heat of oil (S) = 1840 to 2100 J/kg/°C]

Equating this to the amount of artificial cooling required, we have

$$19\,000\ m = 91.4$$

∴ $$m = 91.4 / 19\,000 = 0.0048\ \text{kg / s} = 0.288\ \text{kg / min}$$ **Ans.**

Example *The load on the journal bearing is 150 kN due to turbine shaft of 300 mm diameter running at 1800 r.p.m. Determine the following :*

1. Length of the bearing if the allowable bearing pressure is 1.6 N/mm^2, and

2. Amount of heat to be removed by the lubricant per minute if the bearing temperature is 60°C and viscosity of the oil at 60°C is 0.02 kg/m-s and the bearing clearance is 0.25 mm.

Axle bearing

Solution. Given : $W = 150$ kN $= 150 \times 10^3$ N ; $d = 300$ mm $= 0.3$ m ; $N = 1800$ r.p.m. ; $p = 1.6$ N/mm^2 ; Z = 0.02 kg / m-s ; $c = 0.25$ mm

1. *Length of the bearing*

Let l = Length of the bearing in mm.

We know that projected bearing area,

$$A = l \times d = l \times 300 = 300\, l \text{ mm}^2$$

and allowable bearing pressure (p),

$$1.6 = \frac{W}{A} = \frac{150 \times 10^3}{300\, l} = \frac{500}{l}$$

$\therefore$ $l = 500 / 1.6 = 312.5$ mm **Ans.**

2. *Amount of heat to be removed by the lubricant*

We know that coefficient of friction for the bearing,

$$\mu = \frac{33}{10^8}\left(\frac{Z.N}{p}\right)\left(\frac{d}{c}\right) + k = \frac{33}{10^8}\left(\frac{0.02 \times 1800}{1.6}\right)\left(\frac{300}{0.25}\right) + 0.002$$

$$= 0.009 + 0.002 = 0.011$$

Rubbing velocity,

$$V = \frac{\pi d.N}{60} = \frac{\pi \times 0.3 \times 1800}{60} = 28.3 \text{ m/s}$$

$\therefore$ Amount of heat to be removed by the lubricant,

$$Q_g = \mu.W.V = 0.011 \times 150 \times 10^3 \times 28.3 = 46\,695 \text{ J/s or W}$$

$$= 46.695 \text{ kW } \textbf{Ans.} \qquad \text{... (1 J/s = 1 W)}$$

Example *A full journal bearing of 50 mm diameter and 100 mm long has a bearing pressure of 1.4 N/mm^2. The speed of the journal is 900 r.p.m. and the ratio of journal diameter to the diametral clearance is 1000. The bearing is lubricated with oil whose absolute viscosity at the operating temperature of 75°C may be taken as 0.011 kg/m-s. The room temperature is 35°C. Find : 1. The amount of artificial cooling required, and 2. The mass of the lubricating oil required, if the difference between the outlet and inlet temperature of the oil is 10°C. Take specific heat of the oil as 1850 J / kg / °C.*

Solution. Given : $d = 50$ mm $= 0.05$ m ; $l = 100$ mm $= 0.1$ m; $p = 1.4$ N/mm^2 ; $N = 900$ r.p.m. ; $d / c = 1000$; $Z = 0.011$ kg / m-s ; $t_0 = 75$°C ; $t_a = 35$°C ; $t = 10$°C ; $S = 1850$ J/kg / °C

1. *Amount of artificial cooling required*

We know that the coefficient of friction,

$$\mu = \frac{33}{10^8}\left(\frac{ZN}{p}\right)\left(\frac{d}{c}\right) + k = \frac{33}{10^8}\left(\frac{0.011 \times 900}{1.4}\right)(1000) + 0.002$$

$$= 0.002\,33 + 0.002 = 0.004\,33$$

Load on the bearing,

$$W = p \times d.l = 1.4 \times 50 \times 100 = 7000 \text{ N}$$

and rubbing velocity,

$$V = \frac{\pi d.N}{60} = \frac{\pi \times 0.05 \times 900}{60} = 2.36 \text{ m/s}$$

$\therefore$ Heat generated,

$$Q_g = \mu.W.V = 0.004\,33 \times 7000 \times 2.36 = 71.5 \text{ J/s}$$

Let t_b = Temperature of the bearing surface.

We know that

$$(t_b - t_a) = \frac{1}{2}(t_0 - t_a) = \frac{1}{2}(75 - 35) = 20°\text{C}$$

Since the value of heat dissipation coefficient (C) for unventilated bearing varies from 140 to 420 W/m^2/°C, therefore let us take

$$C = 280 \text{ W/m}^2/° \text{ C}$$

We know that heat dissipated,

$$Q_d = C.A\,(t_b - t_a) = C.l.d\,(t_b - t_a)$$
$$= 280 \times 0.05 \times 0.1 \times 20 = 28 \text{ W} = 28 \text{ J/s}$$

$\therefore$ Amount of artificial cooling required

$$= \text{Heat generated} - \text{Heat dissipated} = Q_g - Q_d$$
$$= 71.5 - 28 = 43.5 \text{ J/s or W } \textbf{Ans.}$$

2. *Mass of the lubricating oil required*

Let m = Mass of the lubricating oil required in kg / s.

We know that heat taken away by the oil,

$$Q_t = m.S.t = m \times 1850 \times 10 = 18\,500\,m \text{ J/s}$$

Finally, equating ;

$$Qt = Q_g - Qd = 43.5 = 18\,500\,m$$

$\therefore$ $m = 43.5/\,18\,500 = 0.00235$ kg / s = 0.141 kg / min **Ans.**

Example *A 150 mm diameter shaft supporting a load of 10 kN has a speed of 1500 r.p.m. The shaft runs in a bearing whose length is 1.5 times the shaft diameter. If the diametral clearance of the bearing is 0.15 mm and the absolute viscosity of the oil at the operating temperature is 0.011 kg/m-s, find the power wasted in friction.*

Solution. Given : d = 150 mm = 0.15 m ; W = 10 kN = 10 000 N ; N = 1500 r.p.m. ; l = 1.5 d ; c = 0.15 mm ; Z = 0.011 kg/m-s

We know that length of bearing,

$$l = 1.5\,d = 1.5 \times 150 = 225 \text{ mm}$$

$\therefore$ Bearing pressure,

$$p = \frac{W}{A} = \frac{W}{l.d} = \frac{10\,000}{225 \times 150} = 0.296 \text{ N/mm}^2$$

We know that coefficient of friction,

$$\mu = \frac{33}{10^8}\left(\frac{ZN}{p}\right)\left(\frac{d}{c}\right) + k = \frac{33}{10^8}\left(\frac{0.011 \times 1500}{0.296}\right)\left(\frac{150}{0.15}\right) + 0.002$$
$$= 0.018 + 0.002 = 0.02$$

and rubbing velocity, $V = \frac{\pi d.N}{60} = \frac{\pi \times 0.15 \times 1500}{60} = 11.78 \text{ m/s}$

** complete the solution as exercise